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TECHNICAL PAPERS

DISCUSSION

ANNOUNCEMENTS

Application of Several Variable-Valve-Timing Concepts to an LHR Engine

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An analytical study was made of advantages provided by electronically controlled hydraulically activated valves when applied to a low heat rejection engine with and without exhaust heat recovery devices. The valves, which could be designed to operate with variable timings and variable rates of opening and closing, would allow the use of certain sophisticated valve strategies not possible with conventional systems. Three individual investigations were carried out. The first concerned the optimization of the timing and rate of valve opening at several speeds and loads, to obtain maximum volumetric efficiency and lowest BSFC. A second investigated early intake valve closing (IVC before BDC), coupled with increased boost, a concept that had been suggested previously in the literature. The present results showed much lower benefits than those predicted by the earlier study. The final study addressed the use of an organic Rankine cycle bottomer (ORCB) to extract energy from the exhaust stream and directing the ORCB output power to the engine air compressor shaft rather than to the engine output shaft. At rated engine conditions, when not employing a power turbine, this concept was found to produce higher BSFC compared to the more standard configuration where the ORBC was directly coupled to the engine shaft. When a power turbine was used between the engine and the ORCB, nearly the same BSFC was achieved with the two configurations.

Introduction

The use of high-temperature thermal barrier materials and coatings in low heat rejection diesel engines presents major long-term opportunities for improved thermal efficiency, reduced package size and weight, due to the elimination of the cooling system, and ultimately longer life and lower cost. In developing an advanced technology low heat rejection engine, it is desirable to evaluate a variety of additional concepts that have the potential to provide synergistic improvements in the areas mentioned above.

One such area concerns the potential use of electronically controlled, hydraulically activated valves (ECV), which would offer total flexibility with respect to timing and considerable flexibility in opening and closing rates. In this paper we describe the results of three studies, in which promising advanced concepts based on such a valve system were evaluated for their potential implementation in low heat rejection diesel engines. One of these was an extensive study of the advantages provided by the ECVs in volumetric efficiency and fuel consumption. Such a concept would provide added flexibility for optimal engine control, by allowing a dynamic adjustment of valve timing and of rate of opening or closing based on instantaneous engine operating conditions. The second study concerned itself with early valve closing (IVC more than 180 crank angle deg before firing TDC), coupled with increased boost, a concept that has been suggested in the literature (Miller, 1946; Chute, 1985). A study was undertaken to evaluate the Miller concept for a turbocharged engine without exhaust heat recovery, in both insulated and cooled configurations. The final study concerned the use of an organic Rankine cycle bottomer (ORCB) to extract energy from the exhaust stream and to use the output power to drive the engine compressor, rather than applying that power directly to the engine shaft. This study also addressed the use of this concept in conjunction with the early intake valve closing, with the ORCB providing additional power to accomplish the requisite high boost.

The evaluation of these three concepts was carried out for the Adiabatic Diesel Reference Engine (ADRE), which is being designed by Detroit Diesel Allison as a part of a DOE/NASA sponsored program (Bennethum and Hakim, 1985).

The primary tool employed in this study was the engine system design analysis code IRIS, described, for example, in Morel et al. (1986). A central feature of this code is a system representation of a turbocharged multicylinder engine, with full transient capability. Among its unique capabilities are highly detailed heat transfer models including convection and radiation from gases to structure, and fully integrated finite element heat conduction, temperature, and stress calculations. All of the models are capable of tracking heat transfer and performance transients during engine speed and load changes.

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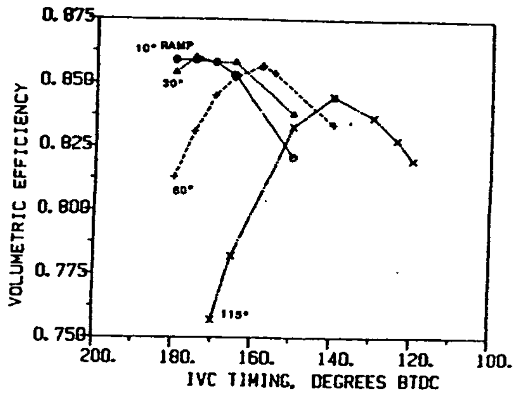

Fig. 1 Variation of volumetric efficiency with intake valve closing angle at various ramp durations; 1800 rpm, fixed plenum conditions

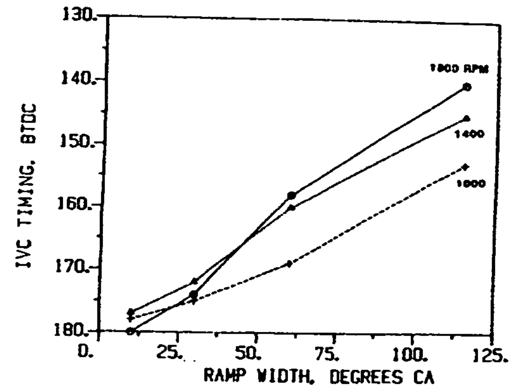

Fig. 2 IVC timing for maximum volumetric efficiency as a function of ramp duration and engine speed (data of Fig. 1)

Electronically Controlled Valve Concept

The electronically controlled valve concept provides additional engine control flexibility by allowing for a variable valve timing as a function of speed and load, or for a given transient condition. The valves, of unit design, would be actuated individually by a camless hydraulic system, which, in addition to variable timing, can also produce a faster rate of valve opening and closing than achievable with conventional cam systems.

A study was carried out to assess the benefits that this flexibility can offer in the following areas: BSFC; pumping losses; volumetric efficiency; power; emissions; cold start; peak pressure control.

The engine analyzed in this study was insulated on the piston top, head, valves, and top of the liner with plasma-sprayed zirconia coating to provide a 62 percent reduction of in-cylinder heat transfer.

The work involved studies of the sensitivity of the engine performance to:

- shape of the opening and closing ramps; these were simulated by sinusoidal curves with variable durations extending from an abrupt 10 deg CA ramp to a gradual 115 deg CA ramp (typical of production cam-driven systems);
- timing of valve opening and closing; searching for the optimum for a particular ramp duration and for a particular engine operating condition.

The engine operating conditions considered were 300 hp and 200 hp power levels at the 1800 rpm rated speed, and 250 hp at the 1200 rpm peak torque speed. The engine was turbocharged with an advanced turbocharger with overall efficiency of 64 percent.

The study was organized in a sequence of steps which constitute analyses of the effects of the individual valve events, and also the interactions between them:

1 intake valve closing;
2 exhaust valve opening;
3 overlap period;
4 interaction of all events at rated conditions, and reoptimization of opening and closing timings at off-rated conditions.

Intake Valve Closing. The evaluation of the intake valve closing event was done with the intake and exhaust plenum conditions and structure temperatures fixed at those levels calculated for the rated conditions with standard valves. The engine was simulated under motoring conditions, and a search was made for the optimum IVC timing for the maximum volumetric efficiency at three engine speeds 1800, 1400, and 1000 rpm. The results for 1800 rpm are shown in Fig. 1. The volumetric efficiency showed a strong sensitivity to timing at

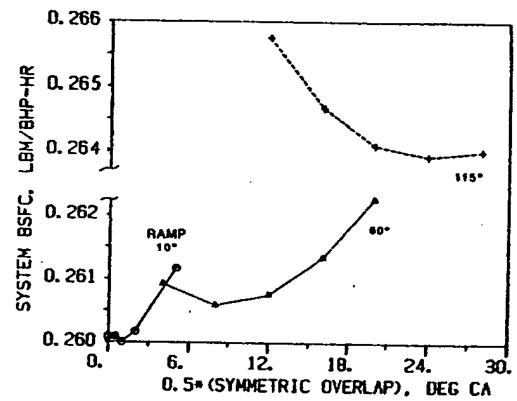

Fig. 3(a) Variations of system BSFC with valve overlap duration at various ramp durations (1800 rpm; full turbocharger simulation)

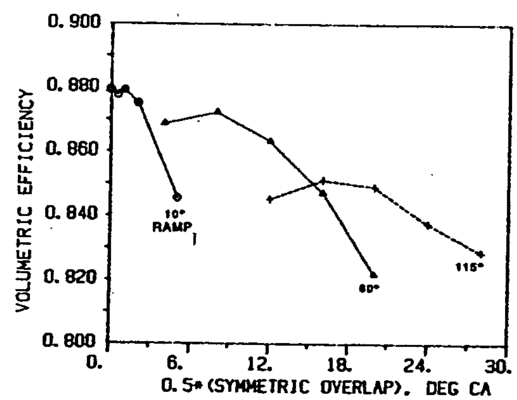

Fig. 3(b) Variation of volumetric efficiency with valve overlap duration at various ramp durations (1800 rpm; full turbocharger simulation)

any fixed ramp duration. By contrast, the maximum values of volumetric efficiency at the various ramp angles varied much less strongly. In fact, there was almost no loss in going from 10 to 60 deg. Increasing the ramp angle beyond 60 deg produced a more substantial loss in η_v, amounting to one percentage point drop at a ramp angle of 115 deg. Plotting the optimum timing angle with respect to ramp duration produces the result shown in Fig. 2. The optimum timing for a valve ramp of short duration is just after BDC, which indicates that the intake valve areas are large enough and that the engine is free-breathing even at the rated speed. This timing is retarded roughly linearly with ramp duration, reaching 30–40 deg CA

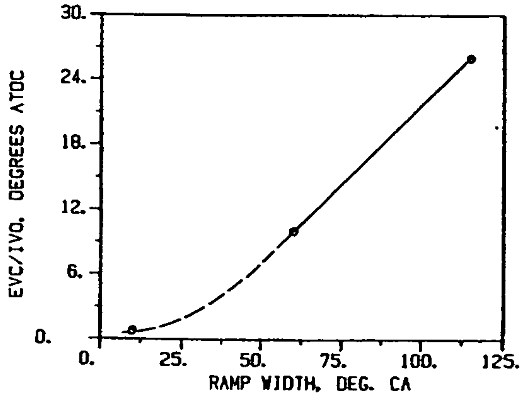

Fig. 4 Half overlap duration for best BSFC versus valve event ramp duration (data of Fig. 3a)

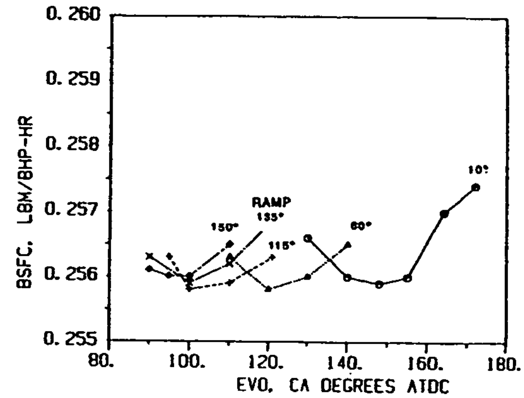

Fig. 6 System BSFC versus exhaust valve opening angle at various ramp durations (1800 rpm; full turbocharger simulation)

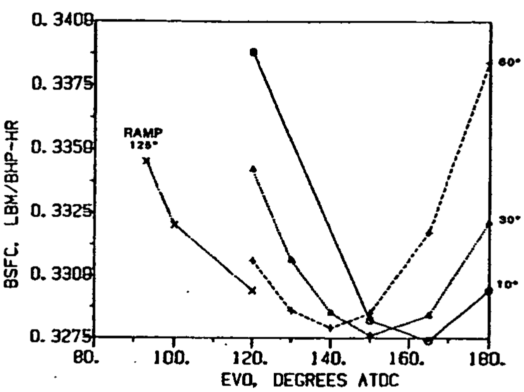

Fig. 5 Reciprocator BSFC versus exhaust valve opening angle at various ramp durations (1800 rpm; fixed plenum conditions)

after BDC (depending on engine speed) at a ramp angle of 115 deg. The figure also shows that the optimum timing depends on engine speed. This means that optimizing the valve timing for rated speed compromises engine breathing somewhat at lower rpm. This is a problem with conventional valves, but it can be eliminated by electronically controlled systems.

IVO/EVC Overlap Period. The IVO/EVC period was studied by simultaneous variation of IVO and of EVC symmetrically about TDC, i.e., in each run the ramp angles of both were equal and the overlap extent was symmetric with respect to TDC. A full turbocharger and exhaust plenum dynamics simulation was used. The monitored parameters were reciprocator BSFC and η_v. The BSFC results showed that best performance is obtained with abrupt valve opening/closing at TDC. However, the degradation with increasing ramp angle was quite small until durations of 60-80 deg, beyond which it became more significant. A similar trend was seen in the volumetric efficiency trends. A series of runs was carried out to determine the optimum overlap for 10, 60 and 115 degree ramps (Fig. 3*a*, *b*). These showed again a relatively small change in BSFC and η_v from 10 to 60 deg ramps, but a significant rise in BSFC beyond that point. The trends of optimum overlap with ramp duration are shown in Fig. 4.

Exhaust Valve Opening. The study of EVO was first carried out with fixed plenum conditions. Again the ramp duration and timing were varied to determine the optimum points. This had almost no effect on volumetric efficiency, and so only BSFC was monitored (Fig. 5). The results showed that at the optimum timing, the reciprocator BSFC was quite independent of ramp duration, with only small degradation with increasing duration. As expected, a large sensitivity of BSFC to EVO at a given ramp duration was observed. To confirm this result for a turbocharged engine, the study was rerun with a full turbocharger simulation and exhaust plenum dynamics. The results showed (Fig. 6) that when the whole system is considered (with exhaust plenum dynamics, turbocharger, power turbine and Rankine cycle bottoming), the sensitivity to ramp duration is even lower; the BSFC at best timing for each ramp was essentially independent of the ramp duration. It also indicated that for conventional valve schedules the EVO timing must be well advanced before the BDC, and that even for a sharp 10 deg ramp it still had to be advanced some 32 deg. This was an interesting result worth further investigation to identify the processes governing the location of the optimum EVO timing. At first glance it would appear that the optimum EVO event should be a sharp ramp with opening at BDC, allowing the full expansion of the combustion gases and thus producing the maximum piston work. Since the optimum timing is not at BDC, this indicates that there is a counteracting effect during the exhaust period which increases as EVO is delayed, eventually more than offsetting the benefits of the additional piston expansion work.

Examination of the detailed plots of exhaust mass flows versus crank angle for the EVO retarded from optimum shows that some of the blowdown takes place after BDC, implying that the piston moves against an elevated pressure on the exhaust stroke. This generates additional pumping work, which increasingly offsets the rapidly diminishing extra piston work obtained by the late EVO timing. It thus appears that the optimum timing is the one which permits the blowdown to be completed within a few degrees after BDC, before the piston starts moving rapidly upward. Since the intensity of the blowdown decreases with decreasing load, and depends also on engine speed, the optimum timing may be expected to vary with these parameters. This variation could be accounted for in an electronically controlled valve system, resulting in a slightly increased engine efficiency.

Interaction of All Valve Events and Comparison to Conventional Valves. In this part of the study, the engine was equipped with power turbine and ORCB, both directly coupled to the engine shaft. Comparisons were made of conventional versus electronically controlled valves. The electronically controlled valves were set to open and close with ramps of 60 deg in duration. This duration was found to be near optimum in the studies of the individual events discussed above. A final optimization was performed by simultaneous variation of all four timings for the 60 deg ramp duration at rated conditions. This optimization resulted in slightly different timings than those detailed in the decoupled studies above, i.e., IVC = −168, EVO = 118, IVO = 352, and EVC = 368. The

Table 1 Engine performance with conventional and electronically controlled valves; engine equipped with power turbine and Rankine cycle bottom

	Rated		Peak torque			Part Load		
	Orig	best 60	Orig	best 60	best 60°*	Orig	best 60	best 60°*
Valve timing IVC	standard	-168	standard	-168	-168	standard	-168	-168
EVO	-	118	-	118	130	-	118	126
IVO	-	368	-	368	368	-	368	360
EVC	-	352	-	352	352	-	352	352
BHP	293.7	297.5	247.2	246.3	247.6	187.3	190.3	190.4
Pumping (psi)	16.2	14.8	1.1	0.4	1.8	14.0	13.2	13.7
Friction (psi)	21.3	20.8	20.3	20.3	20.4	17.4	17.1	17.1
BSFC	0.2586	0.2553	0.2561	0.2563	0.2556	0.2706	0.2663	0.2660
Air-fuel ratio	29.0	29.1	24.5	24.1	24.2	35.2	35.7	35.8
η_v	0.807	0.871	0.861	0.893	0.893	0.803	0.869	0.869
Peak pressure (psi)	2220	2151	2412	2396	2404	1704	1658	1660
Ign. delay (CA)	2.3	2.5	1.6	1.6	1.6	3.0	3.2	3.2
Compressor PR	2.55	2.37	2.47	2.34	2.34	2.04	1.91	1.91
Turbine PR	1.76	1.60	1.64	1.56	1.56	1.60	1.53	1.53
Power turbine PR	1.65	1.66	1.34	1.33	1.33	1.49	1.49	1.49
T, exhaust man. (K)	913	904	951	959	954	809	799	798
NO (ppm)	1634	1640	2372	2371	2373	1106	1116	1114
BSNOX (g/hp-hr)	6.0	5.9	7.1	7.0	6.5	5.1	5.1	5.1
Soot (g/m³)	0.023	0.023	0.013	0.014	0.013	0.019	0.019	0.018
T_{max} avg (K)	1735	1732	1901	1911	1909	1592	1583	1582
T_{max} burned (K)	2853	2857	2890	2890	2890	2793	2802	2802

* Best 60° ramp as determined under peak torque and part load, respectively.

resulting profiles of effective ECV valve areas are shown in Fig. 7(*a*), as compared to the standard valve effective areas shown in Fig. 7(*b*).

The results are summarized in Table 1, which shows comparisons for the three engine operating conditions described above. In all comparisons at a given engine operating condition the fuel flow rate was kept the same. The turbocharger characteristics were adjusted at the rated conditions to produce a desired air fuel ratio for both the original valves and the electronically controlled valves (ECVs). This required compressor pressure ratios of 2.55 and 2.37, respectively. The ECVs produced a higher volumetric efficiency and lower peak firing pressures, while the BSFC was 1.3 percent lower and the predicted emissions were unchanged.

The differences produced by the standard and ECV systems were also observed by comparing the valve mass flow profiles, which displayed increased rate of flow through the electronically controlled exhaust valve during the initial blowdown due to the faster rate of valve opening. They also showed a decrease in the extent of backflows across both the intake and exhaust valves in the ECV system.

For peak torque and part load conditions, Table 1 shows comparison for three cases: standard valves, ECVs with the same timings as at rated conditions, and ECVs with timing schedule reoptimized (but the same ramp duration) for these two engine operating conditions. The turbocharger maps used were the same as used at the rated conditions for the standard valves and ECVs, respectively. Similar results were found as for the rated conditions: The ECVs produced a higher volumetric efficiency, lower peak firing pressures, slightly lower BSFC, and about the same emissions levels.

It was found in this study that it is not necessary to require very fast valve ramps, because the point of diminishing return is reached at a ramp angle of about 60 deg for all three events: IVC, EVO and valve overlap IVO/EVC. This is a significant result from the point of view of the practicality of this concept, as decreasing the valve ramp to very short durations rapidly increases the forces and power required for valve operation.

Comparison of the optimum ECVs to the original conventional valves showed that the ECVs can provide modest benefits in BSFC (on the order of 1.5 percent) and peak firing pressures (on the order of 50 psi) over the coventional valves, which were quite well optimized to begin with. Thus one concludes that the direct application, in freely breathing engines, of electronically controlled valves with sharp opening and closing ramps and *with timings similar to those used in conventional systems* does not produce sufficiently large benefits in volumetric and thermal efficiency to warrant a serious development program in its own right. However, once ECVs

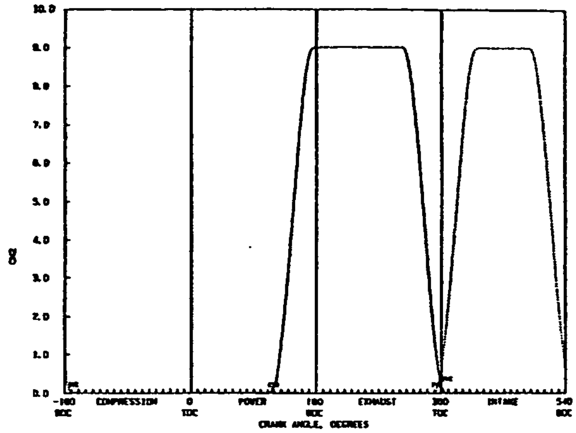

Fig. 7(*a*) Effective valve area profiles for ECVs with 60 deg ramp durations, optimized for operation at rated conditions

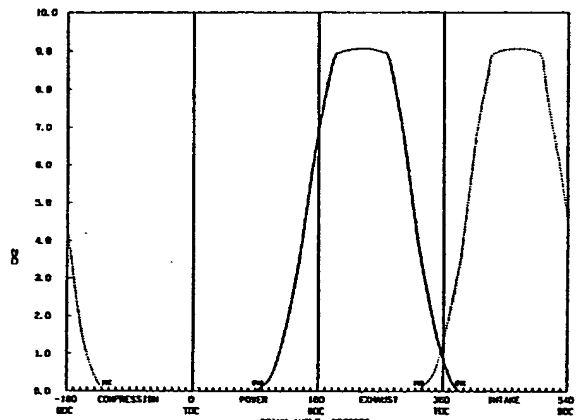

Fig. 7(*b*) Effective valve area profiles of a conventional valve system

are installed on an engine, they permit certain more sophisticated valve timing strategies not possible otherwise:

- reöptimization of valve events with speed and load;
- IVC closing at BDC for cold start and light load operation;
- engine braking by elimination of expansion work, i.e., EVO near TDC.
- selective cylinder cutouts (shutoff valves and injector).

Early Intake Valve Closing

Early intake valve closing (i.e., IVC more than 180 deg before firing TDC) has been suggested in the literature by Miller (1946) as a means to accomplish a larger degree of intercooling and higher BMEP. More recently, the idea was proposed by Chute (1985) as a method of producing higher engine thermal efficiency. The objective of Chute's work was to demonstrate that the energy allowed to be wasted at the exit of turbocharger turbines in today's engines can be used simultaneously to provide additional compressor boost and reduce the engine compression work by early intake valve closing. Chute analyzed the concept and concluded that with highly insulated engines equipped with high-efficiency turbochargers (64 percent overall efficiency), early intake valve closing can produce over 7 percent improvement in BSFC, whereas a much smaller benefit would be available under noninsulated conditions.

Turbocharged Engine With No Exhaust Energy Recovery. In this part of the study the engine studied was

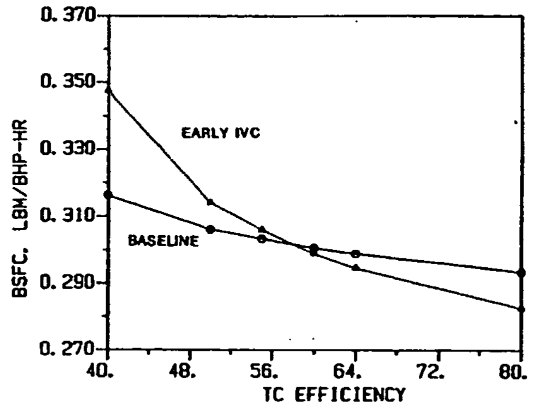

Fig. 8 Fuel consumption as a function of turbocharger efficiency for an ADRE engine: _____ baseline turbocharged engine, --- same engine with early IVC

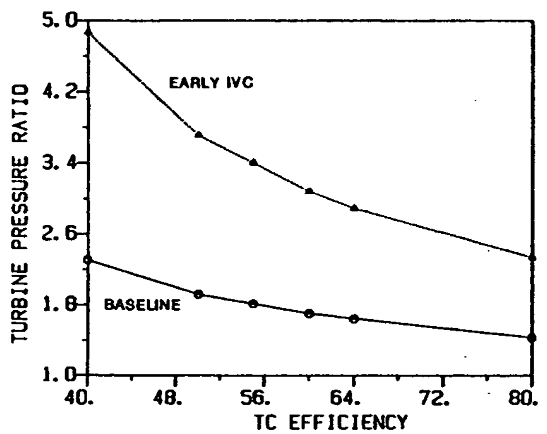

Fig. 9 Turbocharger turbine pressure ratio as a function of turbocharger efficiency for an ADRE engine: _____ baseline turbocharged engine, --- same engine with early IVC

highly insulated, turbocharged, employed no exhaust energy recovery, and had valve ramp durations of 60 deg, except the intake valve closing ramp which was 30 deg in duration for the early-closing case only. The intake valve closed at 168 deg before firing TDC for the baseline engine and at 255 deg for the early-closing case.

The volumetric efficiency declined by almost a factor of two for the early closing case, which required a greatly increased boost so that the fuel air ratio would be unchanged. The air was intercooled to the same level (311 K) for all cases studied, and the impact of this will be discussed subsequently.

The early-closing case was run first with injection timing set so that the combustion start would be the same as in the baseline engine (−7.8 BTDC). A parametric study was then run with variable turbocharger efficiency (turbine × compressor peak efficiencies × mechanical efficiency) over a range from 40 to 80 percent. The results are shown in Fig. 8, which shows that for turbocharger efficiencies greater than about 58 percent the early closing produces lower BSFC than the baseline. The dependence of the early closing concept on turbocharger efficiency is much stronger than for the baseline case. BSFC rises sharply with declining turbo efficiency, and at low turbo efficiencies the early closing is clearly inferior. This trend is tied to the sharp variation in engine backpressure, illustrated in Fig. 9 by the turbine pressure ratio. The compressor pressure ratio was 2.32 for the baseline and 4.26 for the early closing case, and it was independent of the turbo efficiency.

Table 2 Comparison of turbocharged insulated engine with no exhaust heat recovery with conventional valves, and with early closing electronically controlled intake valves; the latter concept incorporates increased boost, produced by a higher backpressure turbocharger turbine; engine at rated fuel rate, 1800 rpm

	Baseline	Early IVC Std BOI	Early IVC Advanced BOI
IVC	-168	-255	-255
EVO	126	126	126
BOI	-10.4	-12.0	-19.0
BHP	254.5	258.0	263.9
Pumping (psi)	-1.8	11.4	13.0
Friction (psi)	20.5	19.9	20.2
BSFC	0.2985	0.2945	0.2878
Air-fuel ratio	29.2	29.2	29.1
ηv	0.896	0.482	0.480
Peak pressure (psi)	2135	1993	2302
Ign. delay (CA)	2.6	4.2	6.3
Comb. start (CA)	-7.8	-7.8	-12.7
T exhaust	864	867	856
T avg. max.	1743	1646	1714
T burned max.	2867	2786	2825
Compressor PR	2.32	4.26	4.26
Turbine PR	1.63	2.89	2.94
NOX (ppm)	1776	1208	1706
BSNOX (g/hp-hr)	7.2	4.9	7.0
Soot (g/m^3)	0.025	0.032	0.020

At an overall turbocharger efficiency of 64 percent, a practical level achievable with advanced turbochargers, the improvement in BSFC was only 1.5 percent. Essentially the same trends were obtained with a *cooled* version of the engine but at higher overall levels of BSFC. Thus, in contrast to the conclusions drawn by Chute, there was no additional advantage generated by engine insulation that would enhance the usefulness of the concept.

One of the very important effects present in the cases studied is the level of intercooling. As already mentioned, in both cases the air exiting the compressor is intercooled to the same level of 311 K, to be accomplished by an air-to-air intercooler. In the baseline case that requires (at compressor efficiency of 80 percent) a reduction of 86 K, equivalent to a heat removal equal to 6 percent of fuel energy. At the same compressor efficiency, the increased compressor pressure ratio and increased compressor outlet temperature of the early IVC closing case lead to the need to reduce the intake air temperature by 174 K, equivalent to a heat removal equal to 12.2 percent of fuel energy. This would place a severe burden on the intercooling system, requiring perhaps an unrealistically large unit. On the positive side, this means that the intake charge starts with the same temperature at IVC, but in the early IVC closing case it is expanded inside the engine cylinder after the valve closes, and so the charge is cooler at the start of compression. As a result the peak pressures are reduced and the lower temperatures lead to lower NO_x emissions as discussed below.

Since the concept showed a measurable predicted benefit in the BSFC, this prompted a more detailed look at the performance at the target level of 64 percent turbo efficiency. The results are shown in Table 2. The first two columns refer to the previously discussed two cases, in which combustion begins at the same crank angle. It may be seen that the early closing case provides not only a decrease in BSFC, but also a reduction in peak firing pressure (by 140 psi) and in NO_x emissions, and there is a small increase in the soot level. This provides an opportunity to achieve additional decrease in BSFC by injection advance. Advancing by the timing to 19 deg BTDC increases the peak pressure to about 170 psi above the baseline and the NO_x emissions are about the same, while the smoke levels are predicted a little lower than the baseline. The BSFC has improved, and is in this cae 3.7 percent lower than the baseline.

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lves, and with early
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at rated fuel rate,

'C	Early IVC Advanced BOI
	-255
	126
	-19.0
	263.9
	13.0
	20.2
45	0.2878
	29.1
2	0.480
	2302
	6.3
	-12.7
	856
	1714
	2825
	4.26
	2.94
	1706
	7.0
2	0.020

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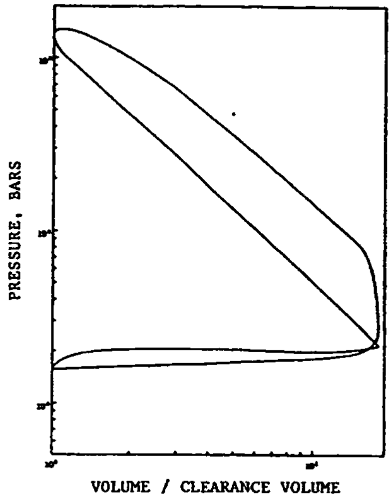

Fig. 10(a) Pressure-volume diagram, ECVs optimized for conventional valve timing schedules

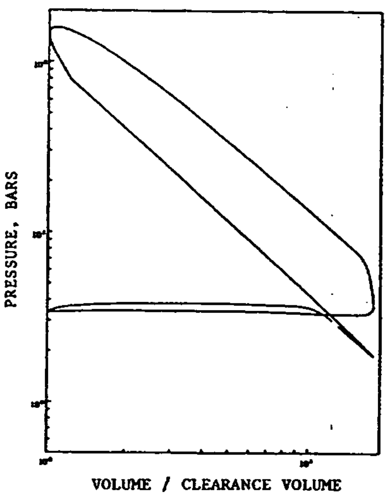

Fig. 10(b) Pressure-volume diagram, ECVs with early intake valve closing

Reducing the injection advance to maintain the same peak pressure, produced a lower BSFC benefit of 3 percent, and also lower NO_x than the baseline.

A more detailed look at the differences between the baseline and the early closing cases is provided by Fig. 10, comparing the pressure-volume diagrams of the two cases, showing clearly the difference near BDC of the intake stroke, where in the early closing case the trapped air is expanded and then compressed again, and also showing the much higher levels of intake and exhaust pressures.

The study also examined whether the IVC timing used (chosen based on Chute's results) was indeed the optimum for this engine and turbocharger efficiency. A range of IVC timings was scanned, again subject to constant A/F ratio and NO_x levels. The results, displayed in Fig. 11, showed that the originally chosen IVC value was very close to the optimum and that no further improvement in BSFC could be obtained at this turbocharger efficiency.

The results obtained disagree with the findings of Chute (1985) in two areas. First, the benefit in BSFC for 64 percent turbo efficiency (the value used by Chute) was only 1.5 percent compared to 7.5 percent found by Chute. Second, while Chute found that engine insulation increased the viability of the con-

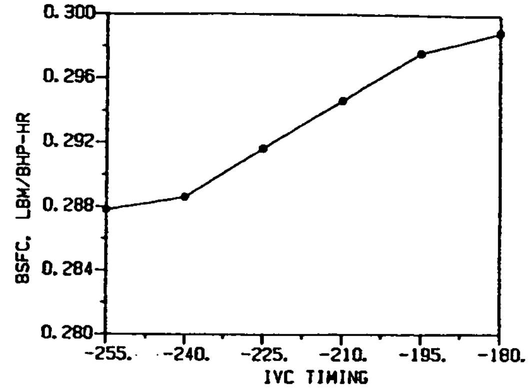

Fig. 11 Fuel consumption for a TC engine with no exhaust he recovery as a function of IVC timing; timing of BOI adjusted for consta NO_x emissions; constant A/F ratio

cept, our results showed no such trends. Another way of look ing at this is to note that at 64 percent efficiency both th present results and Chute's results show that the early intak valve closing provides only small improvement in BSFC fc the cooled engine, if advantage is not taken of the lower NO levels by advancing injection. Further, while our calculation show that this result extends to insulated engines, Chute' show a large improvement with insulation. The difference ca be traced to the heat transfer model, which greatly affects th predicted exhaust temperatures. Chute's exhaust temperature for the adiabatic engines (simulated by setting wa temperatures to unrealistically high values) are sharply highe than for the cooled engine, and this exhaust enthalpy is the available to the turbine for generating the required increase boost. By contrast, the ITI model shows a much smaller ex haust temperature increase with insulation than calculated b Chute and this is responsible for the lower observed benefits

Turbocharged Engine With Exhaust Heat Recover Devices. The above study concerning the early IVC timin was carried out for an engine without heat recovery devices, a conditions similar to those adopted by Chute in his study With that accomplished, the work was extended to an engin configuration which had both a power turbine and ORCB bot tomer. The turbocharger efficiency was fixed at 64 percen and the power turbine efficiency at 78 percent. In all of th runs the valve ramp durations were fixed at 60 deg and th timings were fixed at the optimum values. The injection timin was advanced where appropriate, taking advantage of th lower gas temperature, to a point where peak firing pressure matched those of the baseline case and NO_x emissions wer lower than for the baseline. The A/F ratio was maintained at a constant value of 29:1.

A parametric study was carried out over a range of IVC tim ings seeking the optimum timing for minimum BSFC at rate engine conditions. In contrast to the previous investigation o early IVC for an engine without exhaust heat recovery, thes results showed a very flat BSFC curve with IVC timing (Fig 12*a*), maintaining essentially constant values from −168 t −235 deg before firing TDC. At timings earlier than −23 deg the BSFC started to sharply increase. The compresso pressures required at early IVC timings for maintenance o constant A/F ratio are shown in Fig. 12(*b*).

An analysis of these somewhat surprising results showe that as the IVC timing was advanced, the power produced b the reciprocator kept increasing up to −235 deg much as it di in the earlier study of the engine with no exhaust energ recovery. However, the power produced by the engine exhaus energy recovery devices kept decreasing at about the sam rate, due to the lower energy content of the exhaus

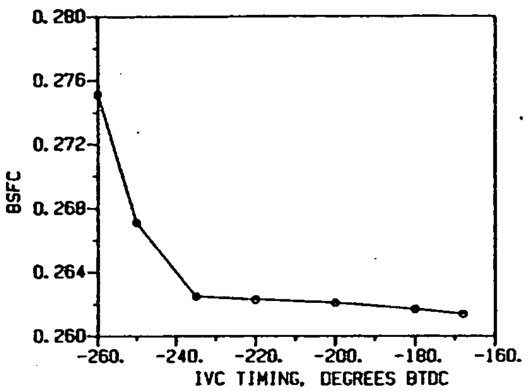

Fig. 12(a) Fuel consumption of the ADRE engine (with power turbine and RCB bottomer) as a function of IVC timing; timing of BOI adjusted for constant peak firing pressure; constant A/F ratio

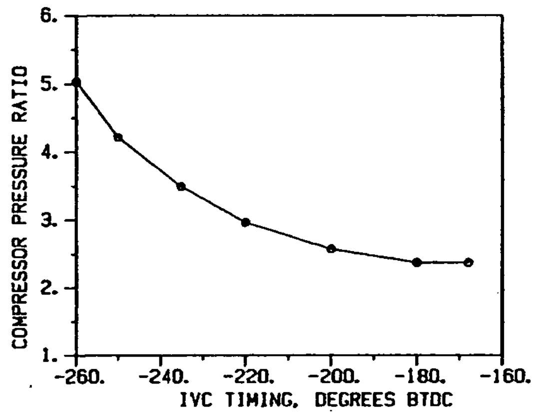

Fig. 12(b) Compressor pressure ratio of the ADRE engine as a function of IVC timing

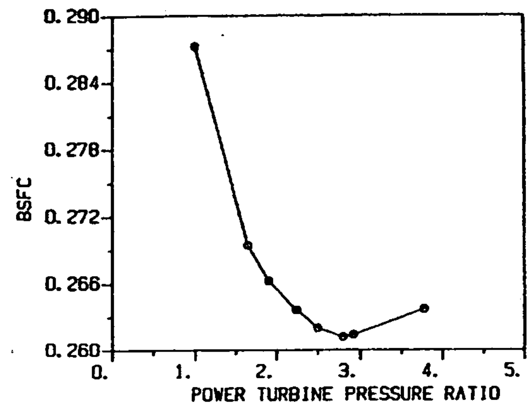

Fig. 13 Effect of power turbine pressure ratio on the fuel consumption of the ADRE engine, with ORCB power used to drive the compressor rather than the engine shaft

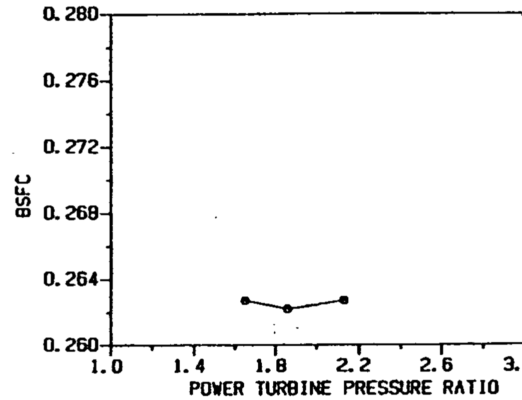

Fig. 14 Effect of power turbine pressure ratio on the fuel consumption of the ADRE with early valve closing (IVC = −235 deg) and ORCB-driven compressor

downstream of the turbocharger turbine. As a result, the early IVC produced no efficiency gain for the engine equipped with an exhaust energy recovery system.

In summary, even with highly efficient turbocharger with 64 percent overall efficiency, and advanced injection timing to take advantage of lower NO_x and lower peak pressures, only a modest 3 percent BSFC improvement was obtained for an engine with no exhaust heat recovery. This conclusion was the same for cooled and insulated engines. The addition of exhaust recovery devices tended to eliminate all of the advantages of the concept.

ORCB-Driven Compressor

One possible concept for insulated engines equipped with ORCB is to use the generated work to compress the air ahead of the turbocharger compressor in an extra stage of compression. This would replace the direct use of the ORCB power to augment the mechanical power produced by the engine (i.e., by gearing it to the engine shaft).

An analysis was carried out employing this idea for an engine equipped with a power turbine placed ahead of the ORCB. This was done both in connection with conventional IVC timing, and in connection with the early intake valve closing concept. The objective was to reduce the power requirement on the turbocharger turbine, thus reducing the backpressure and pumping losses. For the purpose of the simulation, a single-stage turbocharger compressor was used with a common shaft to the turbocharger turbine and the ORCB. The same boost was generated in each case, but the required turbine pressure ratio was less.

ORCB-Driven Compressor With Conventional IVC Timing. At rated conditions, the power available from the ORCB was almost equal to that required by the compressor and so the TC turbine had to produce only a small complementary power. As a result, the engine backpressure was quite low and this provided the opportunity for increasing the power turbine pressure ratio and power. A parametric study was run to determine the optimum power turbine pressure ratio (Fig. 13). The optimum was found to be near PR = 2.8, as a result of opposing trends in reciprocator and power turbine power.

Analysis of the results showed that in this arrangement a large part of the ORCB power is converted through increased pressure work into increased reciprocator power. This is further compounded by the higher exhaust temperature entering the ORCB (due to small pressure drop across the TC turbine), i.e., by increased ORCB power.

The total power produced by the engine at the optimum was almost exactly equal to that of the baseline engine in which the ORCB drives the engine shaft directly. It should be pointed out that this is a more positive result than would be obtained for an engine without a power turbine, where it would be more advantageous to drive the engine shaft directly, rather than driving the compressor. The reason the concept is more favorable when power turbine is used is that there are further

benefits due to increased power turbine work, produced by higher upstream pressure and temperature.

Combined Early IVC and ORCB-Driven Compressor. In this study we combined the two approaches. The calculations were carried out for a single value of IVC timing set at -235 deg, found earlier to be near optimum. It was again necessary to reoptimize the power turbine to take advantage of the reduced TC turbine load. The optimum power turbine pressure ratio was found to lie near 1.86 (Fig. 14). The power produced by the optimum configuration fell slightly short of the power produced by the baseline engine.

In summary, the ORCB-driven compressor, investigated in order to improve the efficiency of the baseline engine, gave at rated conditions results essentially identical to those attainable with the baseline engine with direct coupling of ORCB to the engine shaft. Since there were no advantages in BSFC at steady-state high loads, ORCB-driven compressor would be a favorable alternative only if it could be shown that at low loads or during engine transients there are advantages from BSFC point of view, or for mechanical complexity, or for engine power control reasons. Those aspects were not addressed in this study.

Conclusions

1 For a typical heavy-duty diesel engine equipped with ECVs, it is not necessary to use very fast valve opening/closing ramps, because the point of diminishing return is reached at a ramp angle of 60 deg. Valve ramps of this duration are quite practical from the point of view of the forces and power required for valve operation.

2 In comparison to conventional valve profiles, ECVs provided only modest benefits in BSFC, volumetric efficiency, and peak firing pressures. These benefits are not large enough to warrant the cost of the system on the basis of BSFC gains alone. However, once installed on the engine, they would permit a whole range of certain more sophisticated variable valve timing strategies not possible otherwise, such as high compression cranking, engine braking, cylinder cutouts, volumetric efficiency timing with engine speed, etc.

3 Early intake valve closing (well before BDC), combined with increased boost, can produce BSFC improvements of up to 3 percent, with the same peak firing pressures and lower NO_x. However, a large intercooler is required, removing twice as much energy than in a conventional engine. These conclusions were the same for a cooled, as well as for an insulated engine. When applied to an engine equipped with exhaust heat recovery devices, the concept produced no improvement in BSFC. This was the result of reduced work produced by the exhaust heat recovery devices, which canceled the increased basic engine power output.

4 The observed effects of early valve closing are in disagreement with previous literature. These differences can be traced to the details of models used in evaluation of the concept. The most significant were the differences in the heat transfer models used, with the present models being more physically based.

5 The use of organic Rankine cycle bottomer (ORCB) to drive the intake air compressor produced higher BSFC than when the ORCB was used to drive the engine shaft directly. When an exhaust power turbine driving the engine output shaft was added to the engine, the BSFC levels were equal for the ORCB power directed to either the compressor or the engine output shaft.

Acknowledgments

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